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AN INVESTIGATION OF HEAT TRANSFER ENHANCEMENT AND FRICTION CHARACTERISTICS IN SOLAR AIR HEATER DUCT WITH CUBE SHAPED UNIFORM ROUGHNESS ON THE ABSORBER PLATE

MADHUKESHWARA N[1] & E. S. PRAKASH[2]

[1]Department of Mechanical Engineering, Bapuji Institute of Engineering and Technology, Davanagere, Karnataka, India

[2]Department of Studies in Mechanical Engineering, University B.D.T. College of Engineering, Davanagere, Karnataka, India

ABSTRACT

An experimental investigation of heat transfer enhancement and friction characteristics of fully developed turbulent flow in a rectangular duct of solar air heater with absorber plate having cube shaped uniform artificial roughness on its underside is carried out. The investigation covers wide range of different geometrical parameters of cube shaped uniform roughness. The relative roughness pitch (p/e) used is 10 and 20, relative roughness height (e/D_h) is 0.04, relative roughness gap (w/e) used is 4 and 8 and angle of attack of flow (α) is 90°. Duct aspect ratio (W/B) is kept 5 and Reynolds number (Re) is varied from 3,000 to 8,000. The heat transfer and friction factor values obtained are compared with those of smooth duct under similar flow conditions. Investigation shows 1.82 and 2.87 times enhancement in Nusselt number and friction factor respectively.

KEYWORDS: Friction Factor, Nusselt Number, Thermo-Hydraulic Performance

INTRODUCTION

Solar air heaters absorb the incoming solar radiation and transfer the heat energy to a fluid flowing through the collector. These have found several applications including space heating and crop drying. The efficiency of conversion of incident solar radiation to useful heat energy greatly depends upon effectiveness with which absorber surface transfers the heat to the working fluid. The efficiency of solar air heaters is found to be low because of low convective heat transfer coefficient of the flowing air inside the duct. The absorbed radiation energy by the absorber plate is not completely utilized by the air and leads to energy loss. Fins, artificial roughness on absorber surface and packed beds in the ducts are proposed for the enhancement of thermal performance of the collectors. In this paper cube shaped uniform three dimensional roughness elements on the absorber plate is used for improving the performance of solar air heater. The parameters influencing the heat transfer characteristics are Reynolds number (Re), relative roughness gap (w/e), Relative roughness pitch (p/e), relative roughness height (e/D_h), angle of attack of flow (α) and the aspect ratio (W/B) of the air heater duct. The investigations are carried out to develop a systematic approach for the selection of an optimum design for artificially roughened surface, which improves the heat transfer and reduces the pumping power. Experimental setup is fabricated and tests are conducted to determine the effect of geometrical parameters of cube shaped roughness on heat transfer and friction characteristics. A. R. Jaurker et al. [1] investigated effect of rib-grooved artificial roughness on heat transfer and friction characteristics of solar heater. And from this study it is found that thermo-hydraulic performance of rib-grooved duct is superior as compared to ribbed duct only. Abdul-Malik et al. [2] carried out experimental investigation of the effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air

heater with absorber plate having V-shaped ribs on its underside. They found increase in both Nusselt number and friction factor with increase in roughness height. The second law based analysis of rib roughened solar air heaters is reported by Layek et al. [3]. Atul et al. [5] carried out experimental investigation of heat transfer and friction factor characteristics of rectangular duct roughened with W-shaped ribs on its underside on one broad wall arranged at an inclination with respect to flow direction. D.L. Gee and R.L. Webb [6] contributed experimental information for single phase forced convection in circular tube containing a two dimensional rib roughness. Dhananjay Gupta et al. [7] investigated the effect of relative roughness height, angle of attack on heat transfer and friction factor in rectangular duct having circular wire ribs on the absorber plate. E.M.Sparrow and L.M. Hossfeld [8] conducted experiments to determine the heat transfer, pressure drop and flow field responses to the rounding of the peaks of a corrugated wall duct. J.C. Han [9] revealed from investigations that collector with the fins collected 11% more energy than the collector without fins. But this is accompanied by significant increase in pressure drop. J.C. Han et al. [10] studies effects of heat transfer and friction for rib roughened surfaces. Correlation for friction factor and heat transfer is developed to account for rib shape, spacing and angle of attack. M.J. Lewis [11] carried out an elementary analysis for predicting the momentum and heat transfer characteristics of a hydraulically rough surface. In this study equally spaced rectangular ribs are considered and the flow is represented by a series of attached and separated flow regions. M. M. Sahu and Bhagoria [12] have investigated the effect of 90^{o} broken wire ribs on heat transfer coefficient of a solar air heater duct. A pitch of 20 mm gives the highest thermal efficiency of 83.5% for and element height of 1.5 mm and reported heat transfer coefficient of roughened duct is 1.25 to 1.4 times compared to smooth duct under similar operating conditions at higher Reynolds number. M. K. Gupta et al. [13] investigated the effect of relative roughness height, angle of attack on heat transfer and friction factor in rectangular duct having circular wire ribs on the absorber plate. M. K. Mittal et al. [14] compared the effective efficiency of the solar air heater having different types of roughness elements on the absorber plate. N. Sheriff and P. Gumley [15] investigated experimentally the heat transfer and friction characteristics of a surface with discrete roughness. R.L. Webb and E.R.G. Eckert [16] conducted a comparative study between the roughened tubes and smooth tubes in design of heat exchangers. This study is conducted mainly to achieve enhancement of heat transfers capacity and also to reduce the friction factor. Webb et al. [17] developed heat transfer and friction correlations for turbulent flow in tubes having repeated rib roughness. The friction correlation is based on “Law of the wall similarity”. R.P. Saini and Jitendra Verma [18] developed heat transfer and friction factor correlations for a duct having dimple shape artificial roughness on the underside of the absorber surface. Santosh B. Bopche and Madhukar S. Tandale [19] investigated heat transfer and frictional characteristics of a turbulator roughened solar air heater duct. Enhancement in the heat transfer and friction factor by 2.82 and 3.72 times, are reported respectively in the turbulator roughened duct. Sukhmeet Singh et al. [20] carried out exergy based analysis of solar air heater having discrete V-down rib roughness on absorber plate. The present investigation is, therefore, carried out with the objective of extensive experimentation in order to increase the heat transfer in solar air heater ducts with artificial roughness is the form of three dimensional uniform roughnesses i.e. cube shaped roughness elements on the absorber plate.

EXPERIMENTAL PROGRAM

Experimental Setup

A schematic diagram of experimental setup is shown in Figure 1. The flow system consists of smooth entrance section, roughened entrance section, test section, an exit section, mixing chamber, a flow meter and an air blower. The duct is of size 2600mm × 150mm × 30mm made of wooden panels. Test section is of length 1200mm. The length of smooth entrance section and exit section are 800mm and 600mm respectively. For a roughened duct thermally fully developed flow will be established in a short length i.e 2–3 times of hydraulic diameter therefore short entrance length is chosen. For the turbulent flow regime, the lengths are considered as per ASHRAE standard 93–97 [4]. A 1.626 mm

thick aluminium sheet of 1200mm × 150mm size is used as an absorber plate and the lower surface of the plate provided with cube shaped artificial roughness with various values of relative roughness pitch and relatve roughness gap. Relative roughness pitch of the roughness elements is varied from 10 to 20 and relative roughness gap is varied from 4 to 8. An electric heater is used to provide a uniform heat flux up to a maximum of 1500 W/m^2 to the absorber plate. The power supply to the heater is controlled through an AC variac. Uniform cube shaped roughness elements are pasted below the absorber plate to create artificial roughness. The flow rate of air in the duct is measured using an orifice meter connected to U–tube manometer. The duct is insulated from the three sides to ensure that all the heat flux which is supplied from the heater plate is transferred to the duct and also to minimize the losses to the surroundings. The other end of the duct is connected to a circular pipe via a rectangular to circular transition section. Calibrated chromel–alumel thermocouples are used to measure air and plate temperatures at various locations. The pressure drop in the test section is measured using a micromanometer.

Instrumentations

Temperature Measurement

Calibrated chromel–alumel thermocouples with digital temperature indicator, indicating output in degree centigrade with an accuracy of 0.1 ^{o}C through a selector switch are used to measure temperatures of air and absorber plate at various locations. Eight thermocouples are pasted on the top surface of the absorber plate to record the plate temperature. The locations of thermocouples on absorber plate are shown in Figure2(a). The temperature of air inside the duct is recorded by ten thermocouples inserted inside the duct at different locations as shown in Figure 2(b). Also inlet and outlet temperatures of air are measured by two separate thermocouples.

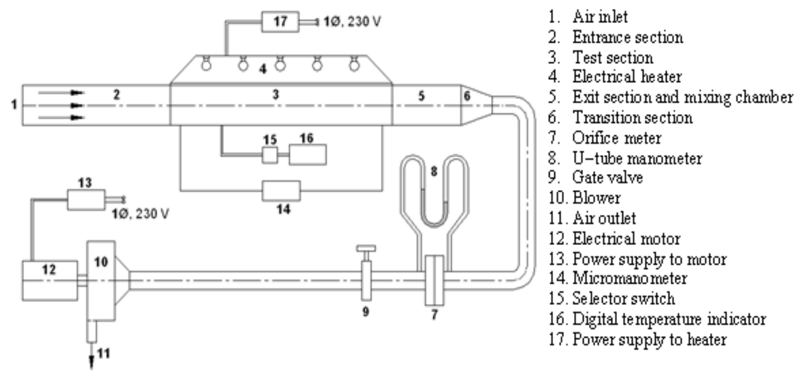

Figure 1: Schematic Diagram of Experimental Setup

Air Flow Measurement

The air is sucked through the rectangular duct by means of a blower of discharge 3.3 m^3/min. It is driven by a single phase, 230 V, 600 W and 16,000 r.p.m., AC motor. Control valve is used to control the air flow rate through the duct. The flow rate of air in the duct is measured using orifice meter connected to an U tube manometer.

Pressure Drop Measurement in the Duct

The pressure drop in the test section is measured by a micro manometer which consists of a relatively large reservoir of fluid connected through a flexible hose to a glass tube that is inclined five degrees from the horizontal. Inclined glass tube is mounted on a sliding arrangement with a screw having a pitch of 1.57mm and a graduated dial having 360 divisions, each division showing a movement of 0.00436 mm of the reservoir. The meniscus of the liquid in glass tube is maintained at a fixed prescribed mark by sliding the tube up or down. The movement is noted which yield the pressure difference across the test length. Diesel having specific gravity 0.8373 is used in the manometer to increase

the accuracy further.

Roughness Geometry and Range of Parameters

Cube shaped roughness geometry on the absorber plate are arranged in the inline and staggered manner. The range of parameters covered under the present experimental study is as follows:

- Reynolds number $3000 \leq Re \leq 8000$.
- Relative roughness pitch $10 \leq p/e \leq 20$.
- Relative roughness gap $4 \leq p/e \leq 8$.
- Relative roughness height $e/D_h = 0.04$
- Angle of attack of flow $\alpha = 90^o$.

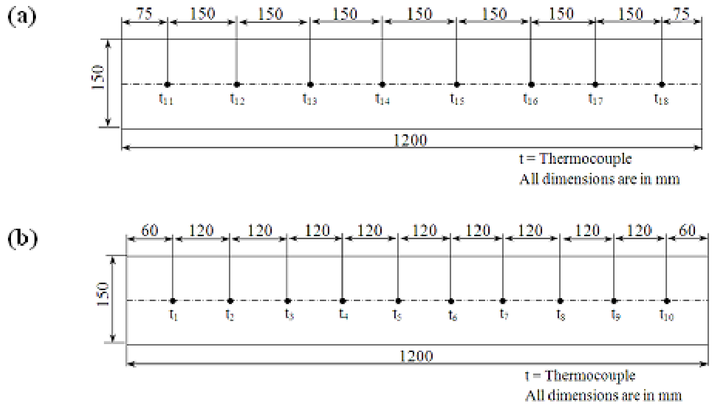

Figure 2: (a) Position of the Thermocouples on the Absorbing Plate (Test Length)
(b) Position of the Thermocouples in the Air Duct (Test Length)

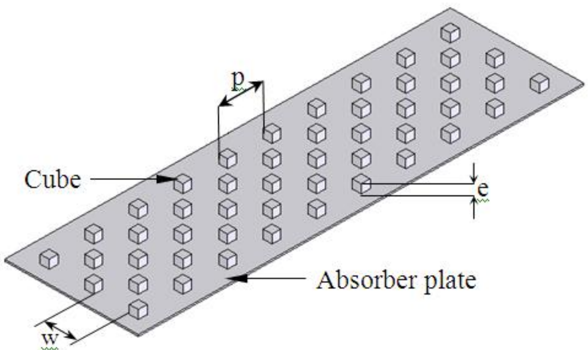

Figure 3: Schematic Diagram of Staggered Arrangement of Cube Shaped Roughness Geometry on the Absorber Surface

EXPERIMENTAL PROCEDURE

Before starting the experiment all the components of the setup and instruments viz. micro–manometer, U–tube manometer, voltmeter, ammeter and thermocouples are checked carefully for their proper operation. Ribbed absorber plate is installed and the test section is assembled. The energy for heating is supplied by the heater for one hour to the test section. After one hour the blower is then switched on ensuring no leakage in the joints of the duct and pressure tappings. The mass flow rate in the duct is adjusted using control valves. Once the mass flow rate of air is fixed, all the readings are taken when the system attains quasi–steady state. Quasi–steady state means that the temperatures do not change appreciably for 10–15 min. It is decided to conduct tests for eleven values of mass flow rate of air in order to

cover the entire range of Reynolds number. The following parameters are measured during experimentation:

- Temperature of the absorber plate at various locations.
- Temperature of the air at inlet, outlet and also at various locations of the test section.
- Pressure difference across the orifice meter.
- Pressure drop across the test section.
- Voltage and current supply to the heater.

DATA REDUCTION

Steady state values of the plate and air temperatures in the duct at various locations are obtained for a given heat flux and mass flow rate of air. Nusselt number and friction factor are also calculated to know the effect of roughness geometry and operating parameters on heat transfer and friction characteristics. The following expressions are used for the calculation of mass flow rate (m), velocity of air (V), heat supplied to the air (q), and heat transfer coefficient (h).

$$m = \rho C_d a_o \sqrt{2gH/(1-(a_o / a_1)^2)} \tag{1}$$

The calibration of orifice plate is done against a standard pitot tube which gives a value of 0.62 for coefficient of discharge (C_d). Where, $H = h_m(\rho_w/\rho)$ and (h_m) is manometer reading. The velocity of air in the duct is calculated as follows.

$$V = m/\rho WB \tag{2}$$

Useful heat gain to the air flowing in the duct is calculated as follows.

$$q = mC_p(t_o - t_i) \tag{3}$$

The temperature t_o and t_i are values of the air inlet and outlet temperatures respectively. Convective heat transfer coefficient for the test section is calculated as follows.

$$h = q/(A_c\,(t_p - t_f)) \tag{4}$$

Where, (A_c) is the area of absorber plate, the temperature (t_p) and (t_f) are average values of the absorber plate and fluid (air) temperature, respectively. The convective heat transfer coefficient is then used to obtain the average Nusselt number from the following expression.

$$Nu\_av = (hD_h)/k \tag{5}$$

The friction factor is determined from the measured values of pressure drop across the test length, L = 1200mm using following expression.

$$f_r = (D_h\,\Delta p)/(2LV^2\rho) \tag{6}$$

The values of Nusselt number and friction factor are determined for smooth duct and compared with the values obtained from correlations of modified Dittus–Boetler equation for the Nusselt number and modified Blasius equation for the friction factor. Nusselt number for a smooth rectangular duct is given by modified Dittus–Boetler equation as,

$$Nu_s = 0.023\, Re^{0.8}\, Pr^{0.4}\, (2R_{av}/D_h)^{-0.2} \quad \text{Where, } 2R_{av}/D_h = (1.156 + H/W - 1)/(H/W) \tag{7}$$

Friction factor for a smooth rectangular duct is given by the modified Blasius equation as,

$$f_s = 0.085\ Re^{-0.25} \tag{8}$$

Average friction factor for three smooth sided and one (top) rough sided wall is calculated using following well known equation.

$$f_{av} = ((W/B + 2)\ f_r + (W/B)\ f_s)/2(W/B + 1) \tag{9}$$

An important parameter called efficiency index (η) is defined as ratio of enhancement factor of heat transfer coefficient to that of friction coefficient. [17]

$$\eta\ =\ (St_{av}/St_s\ )/(f_{av}/f_s) = (Nu_{av}/Nu_s\ )/(f_{av}/f_s) \tag{10}$$

ERROR ANALYSIS

An error analysis of experimental measurements is carried out on the basis of method proposed by Kline and McClintock [21]. This method is used for the prediction of uncertainty which should be associated with an experimental result based on observations of the scatter in the raw data used in calculating the result. The maximum possible measurement errors in the values of major parameters of present investigation are given below.

- Reynolds number = ± 1.66%
- Heat transfer coefficient = ± 4.60%
- Nusselt number = ± 4.60%
- Stanton number = ± 4.89%
- Friction factor = ± 2.83%

RESULTS AND DISCUSSIONS

The values of Nusselt number and friction factor for cube shaped roughness geometry on the absorber surface are computed on the basis of experimental data collected for various flow and roughness parameters. The effects of various parameters on Nusselt number and friction factor are presented in this section. Variation of absorber plate temperature and fluid temperature in the duct is shown in Figure4. The temperatures t_p and t_f are the average values of absorber plate temperature and fluid temperature respectively. The average value of plate temperature is determined from eight thermocouples readings at different locations on the absorber plate as shown in Figure2 (a). The average value of fluid temperature is found from the readings of ten thermocouples fixed in central locations of the duct cross-section along the flow direction as shown in Figure 2(b). It is found that in the span wise direction the variation of temperature is negligible. This variation of temperature is compared with work of Abdul-Malik et al. [2].

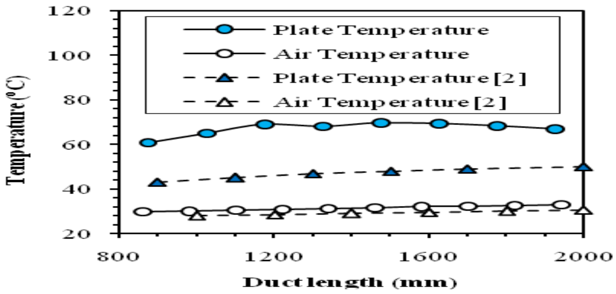

Figure 4: Typical Variations of Plate and Air Temperature along the Length of the Test Section

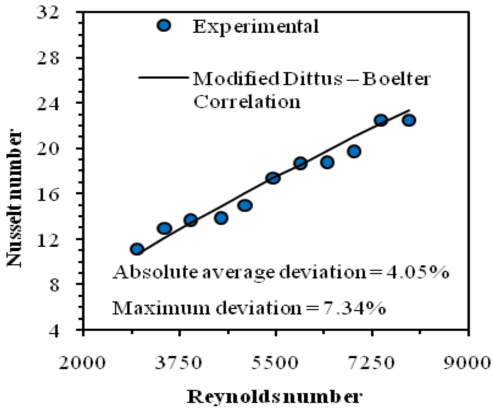

Figure 5: Comparison of Experimental Values and Predicted Values of Nusselt Number for Smooth Duct

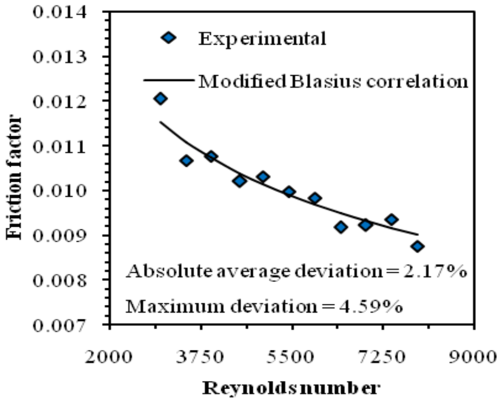

Figure 6: Comparison of Experimental Values and Predicted Values of Friction Factor for Smooth Duct

Comparison of experimental and predicted values of Nusselt number and friction factor are shown in Figures 5 and 6. A good agreement between the experimental and theoretical values ensures accuracy of the experimental setup. Nusselt number and friction factor of smooth duct proposed by modified Dittus Boelter correlation and modified Blasius Correlation resepectively.

Effect of Reynolds Number

Figure 7 and Figure 8 illustrates the effect of Reynolds number on Nusselt number and friction factor respectively for different configurations of the absorber plate. It is observed that Nusselt number increases and friction factor decreases with increase in Reynolds number. The values of Nusselt number and friction factor for roughened absorber plate are considerably higher as compared to smooth one for a given Reynolds number. This is due to a distinct change in the fluid flow characteristics as a result of roughness that causes flow separations, reattachments and the generation of secondary flows.

Effect of Relative Roughness Pitch

From Figure 7 and Figure 8 it is clear that Nusselt number and friction factor values are higher for relative roughness pitch (p/e) value of 10. Relative roughness pitch is defined as the ratio of distance between two consecutive roughness elements to the height of the roughness element.

Nusselt number and friction factor decreases with increase in relative roughness pitch (p/e).This is because with the increase in relative roughness pitch, number of reattachment points over the absorber plate reduces. Therefore, relative roughness pitch value should be taken less so that number of reattachment points increases and augmentation in heat transfer takes place but it is seen that the value of relative roughness pitch should not be less than eight [17], in this case

reattachment point could not form. It is observed that maximum value of Nusselt number is obtained for relative roughness pitch (p/e) value of 10. Also many investigations revealed that reattachment does not occur for relative roughness (p/e) value less than about 8 to 10.

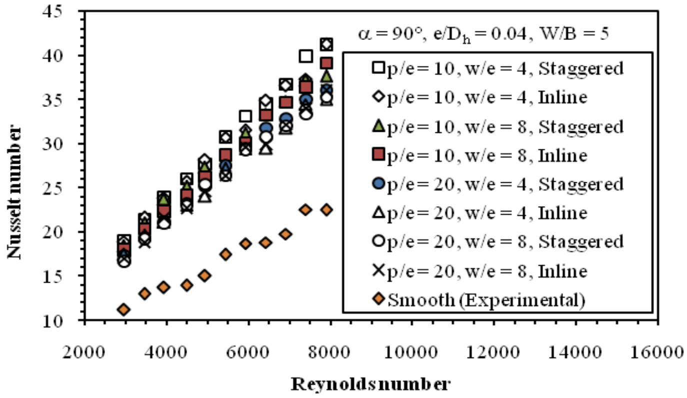

Figure 7: Variation of Nusselt Number with Reynolds Number for Different Configurations of the Absorber Plate

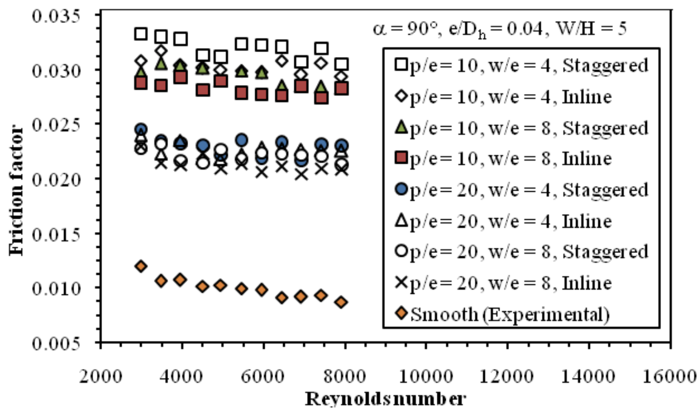

Figure 8: Variation of Friction Factor with Reynolds Number for Different Configurations of the Absorber Plate

Effect of Relative Roughness Gap

From Figure 7 and Figure 8 it is very much clear that Nusselt number and friction factor values are higher for relative roughness gap (w/e) value of 4. This is due to the fact that lesser value of relative roughness gap produces more turbulence because of the more number of roughness elements on the absorber surface. Also it is observed that for a considered value of relative roughness pitch (p/e) and relative roughness gap (w/e) staggered arrangement gives higher values of Nusselt number and friction factor compared to inline arrangement of roughness elements.

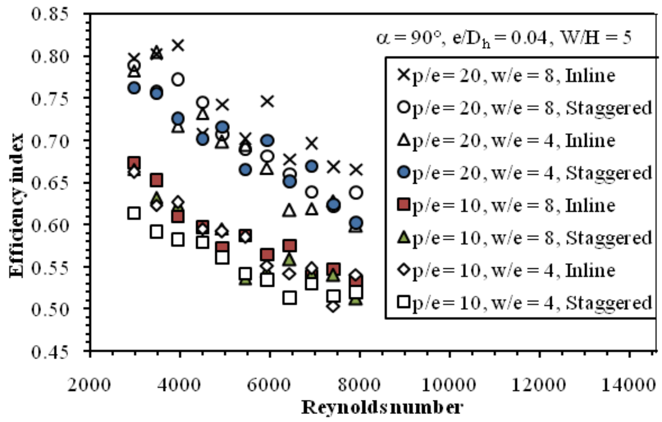

Figure 9: Variation of Efficiency Index with Reynolds Number for Different Configurations of the Absorber Plate

Thermohydraulic Performance

An important thermohydraulic performance parameter called efficiency index (η) is defined as ratio of enhancement factor of heat transfer coefficient to that of friction coefficient. It is found that the artificial roughness on the absorber plate of the roughened duct results in considerable enhancement of heat transfer.

This enhancement is, however, accompanied by a substantial increase in friction factor. It is, therefore, desirable that to select the roughness geometry such that the heat transfer is maximized while keeping the friction losses at the minimum possible value. This requirement of the collector can be fulfilled by considering the heat transfer and friction characteristics simultaneously.

A parameter called efficiency index that facilitates the simultaneous consideration of thermal and hydraulic performance is given by R.L. Webb. et. al. [17]. Efficiency index is plotted in Figure 9 against Reynolds number for relative roughness pitch (p/e) of 10 and 20, relative roughness pitch (w/e) of 4 and 8, relative roughness height (e/D_h) of 0.04, angles of attack (α) of 90^o and for staggered and inline arrangements. Highest value of efficiency index can be observed for inline arrangement of roughness elements with relative roughness pitch (p/e) of 20 and relative roughness gap (w/e) of 4 for fixed values of relative roughness height (e/D_h) of 0.04, angles of attack (α) of 90^o and aspect ratio (W/B) of 5.

CONCLUSIONS

An extensive investigation is carried out with an intention to develop a systematic approach for the selection of an optimum design for cube shaped uniform roughness surface, which improves heat transfer characteristics as well as reduces the pumping power requirement.

The following conclusions can be drawn from this work:

- In general, Nusselt number increases and friction factor decreases with an increase of Reynolds number. Nusselt number and friction factor are significantly higher as compared to those obtained for smooth absorber plates. The maximum value of Nusselt number and friction factor is found corresponds to relative roughness pitch (p/e) of 10 and relative roughness gap (w/e) 4 for staggered arrangement of roughness elements on plate. The minimum value of Nusselt number and friction factor is found corresponds to relative roughness pitch (p/e) of 20 and relative roughness gap (w/e) 8 for inline arrangement of roughness elements on plate. It is therefore, roughness parameters of the geometry can be selected by considering net heat gain and corresponding pumping power required to drive the air through the duct.
- In this study it is shown that pumping power required to force the fluid for same heat transfer surface and fluid temperature difference, will be minimum when, $(f_{av}/f_s) < (Nu_{av}/Nu_s)$. This shows any increase in the friction factor increases the heat transfer characteristics of roughened surface resulting in a more efficient heat transfer surface. Highest value of efficiency index can be observed for inline arrangement of roughness elements with relative roughness pitch (p/e) of 20 and relative roughness gap (w/e) of 8 for fixed values of relative roughness height (e/D_h) of 0.04, angles of attack (α) of 90^o and aspect ratio (W/B) of 5.
- The maximum enhancement in Nusselt number and friction factor values compared to smooth duct are of the order of 1.82 and 2.87 respectively. It can therefore be concluded that the roughened absorber solar air heater performs better than that of a smooth absorber air heater.

Nomenclature

A_c	Area of absorber plate (m^2)
a_o	Area of the orifice (m^2)
a_1	Area of the pipe (m^2)
B	Height of the duct (m)
C_p	Specific heat of air (J/kg K)
C_d	Coefficient of discharge (dimensionless)
d_o	Diameter of the orifice (m)
d_1	Diameter of the pipe (m)
D_h	Hydraulic diameter of duct (m)
e	Height of roughness element (m)
e/D_h	Relative roughness height
f_{av}	Average friction factor
f_s	Friction factor of smooth duct
g	Acceleration due to gravity (m^2/s)
h	Heat transfer coefficient (W/m^2 K)
h_m	Manometer reading (m)
k	Thermal conductivity of air (W/m K)
L	Test length (m)
m	Mass flow rate of air (kg/s)
Nu_{av}	Average Nusselt number
Nu_s	Nusselt number of smooth duct
p	Rib pitch (m)
p/e	Relative roughness pitch
Pr	Prandtl number
q	Rate of heat transfer to air (W)
Re	Reynolds number
St_{av}	Average Stanton number
St_s	Stanton number of smooth duct
t_i	Air inlet temperature (^oC)
t_o	Air outlet temperature (^oC)

t_f	Average temperature of fluid (^oC)
t_p	Average temperature of plate (^oC)
V	Velocity of air in the duct (m/s)
W	Width of the duct (m)
w	Gap between roughness elements
W/B	Aspect ratio
w/e	Relative roughness gap

Greek Symbols

α	Angle of attack of flow (^{o})
Δp	Pressure drop in the test length (N/m^2)
ρ	Density of air (kg/m^3)
ρ_w	Density of water (kg/m^3)

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